# Importance of exergy for analysis, improvement, design, and assessment



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This study discusses what exergy analysis brings new to the system design, analysis, assessment, and improvement of energy systems. It also presents how efficiencies are defined for performance assessment. Case studies are given to explain how system exergy analysis is performed, and its efficiencies are evaluated for comparison. Exergy destruction rates for each component of the systems are also studied and presented to investigate the possibilities for determining the magnitudes and improving the performance. © 2012 John Wiley & Sons, Ltd.

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## INTRODUCTION

In essence thermodynamics is defined as the science of energy and entropy, referring to the first and second law of thermodynamics. We further elaborate on it and redefine it as the science of energy and exergy. This will make it thermodynamically more correct because both energy and exergy quantities possess the same units and the efficiency can be defined from both energy and exergy points of views.

Exergy analysis is a thermodynamic analysis technique based primarily on the second law of thermodynamics and appears to be the only tool for assessing and comparing processes and systems rationally and meaningfully. Consequently, exergy analysis can assist in improving and optimizing designs and analyses. Two key features of exergy analysis are (1) it yields efficiencies that provide a true measure of how nearly actual performance approaches the ideal, and (2) it identifies more clearly than energy analysis the types, causes, and locations of thermodynamic losses.<sup>1</sup>

In this article, we exemplify the crucial role of exergy and its use for analysis, improvement, design, and assessment (AIDA). Its linkages to environmental impact and sustainability are also discussed. Fundamental aspects of exergy analysis are further treated and connected to practical systems and applications. Then, two case studies are assessed involving the application of exergy analysis to integrated energy systems, including biomass, geothermal, and steam power systems. Operating parameters such as ambient temperature is varied to investigate its effects on the energy and exergy efficiencies of the overall systems studied. Also, exergy destruction rates for individual components of an integrated system are also studied and presented to determine the magnitudes and possibilities for performance improvement.

## A NEW CONCEPT: AIDA

When we deal with processes, systems, applications, and so on, there are four key items, namely analysis, improvement, design and assessment, as one needs to address. In conjunction with this, we introduce a new AIDA concept to cover these four items. Aida is a female name and originally comes from Arabic.<sup>2</sup> It has many meanings from being happy to distinguishing and from helper to visitor. Because these four letters make this name, we are inclined to take the meaning of helper and make the connection to exergy. Exergy is really a thermodynamic tool for AIDA as clearly illustrated in Figure 1.

Exergy analysis permits many of the shortcomings of energy analysis to be overcome. Exergy analysis, by stemming from the second law of thermodynamics, is useful in identifying the causes, locations,

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FIGURE 1 Exergy for AIDA.

and magnitudes of process inefficiencies. The exergy associated with an energy quantity is a quantitative assessment of its usefulness or quality. Exergy analysis acknowledges that although energy cannot be created or destroyed, it can be degraded in quality, eventually reaching a state in which it is in complete equilibrium with the surroundings and hence of no further use for performing tasks.

Recently, exergy has been a prime tool and its use has been extended to economy under exergoeconomics (or thermoeconomics) by including cost accounting; environment under exergoenviromics (or exergoenvironmental analysis) by including environmental impact accounting (assessment); and exergosustainability by including sustainability accounting (assessment).

## ENERGY AND EXERGY ANALYSES

Energy and exergy fundamentals are described in this section, and methods for applying exergy analysis are discussed. Although a relatively standard terminology and nomenclature has evolved for conventional classical thermodynamics, there is at present no generally agreed upon terminology and nomenclature for exergy analysis. A diversity of symbols and names exist for basic and derived quantities.<sup>3,4</sup> For example, exergy is often called available energy, availability, work capability, essergy, and so on; and exergy consumption is often called irreversibility, lost work, dissipated work, dissipation, and so on. The exergy analysis nomenclature used here follows that proposed by Ref 3 as a standard exergy nomenclature.

## Balance Equations for Mass, Energy, Entropy, and Exergy

A general balance for a quantity in a system may be written as

Inputs - Outputs = Accumulation(1)

where, input and output refer, respectively, to quantities entering and exiting through system boundaries. Generation and consumption refer respectively to quantities produced and consumed within the system. The generation term becomes an input term, whereas the consumption is an output term. Accumulation refers to buildup (either positive or negative) of the quantity within the system.

Versions of the general balance equation as given above may be written for mass, energy, entropy, and exergy. Mass and energy, being subject to conservation laws (neglecting nuclear reactions), can be neither generated nor consumed. Consequently, the general balance (Eq. 1 for each of these quantities becomes

Energy input – Energy output = Energy accumulation (3)

(Entropy input + Entropy generation)

-Entropy output = Entropy accumulation(4)

Exergy input – (Exergy output

Here, entropy is created during a process due to irreversibilities, but cannot be consumed. However, exergy is consumed during the process due to destructions and losses as to be taken into consideration. These balances describe what is happening in a system between two instants of time. For a complete cyclic process where the initial and final states of the system are identical, the accumulation terms in all the balances are zero.

#### **Exergy of Systems and Flows**

Two types of systems are normally considered: open (flow) and closed (nonflow). In general, open systems have mass, heat and work interactions, and closed systems have heat and work interactions. Mass flow into, heat transfer into and work transfer out of a system are defined to be positive. Mathematical formulations of the principles of mass and energy conservation and entropy nonconservation can be written for any system, following the general physical interpretations in Eqs 2–5.

### Exergy of a Closed System

The exergy  $Ex_{nonflow}$  of a closed system of mass *m*, or the nonflow exergy, is defined as

$$Ex_{nonflow} = Ex_{nonflow,ph} + Ex_0 + Ex_{kin} + Ex_{pot}$$
(6)

where

$$Ex_{pot} = PE$$

$$Ex_{kin} = KE$$

$$Ex_0 = \sum_i (\mu_{i0} - \mu_{i00}) N_i$$

 $Ex_{nonflow,ph} = (U - U_0) + P_0(V - V_0) - T_0(S - S_0)$ 

where the system has a temperature *T*, pressure *P*, chemical potential  $\mu_i$  for species *i*, entropy *S*, energy *E*, volume *V*, and number of moles  $N_i$  of species *i*. The system is within a conceptual environment in an equilibrium state with intensive properties  $T_0$ ,  $P_0$ , and  $\mu_{i00}$ . The quantity  $\mu_{i0}$  denotes the value of  $\mu$  at the environmental state (i.e., at  $T_0$  and  $P_0$ ). The terms on the right side of Eq 6 represent respectively physical, chemical, kinetic, and potential components of the nonflow exergy of the system.

Exergy is a property of the system and conceptual environment, combining the extensive properties of the system with the intensive properties of the environment.

Physical nonflow exergy is the maximum work obtainable from a system as it is brought to the environmental state (i.e., to thermal and mechanical equilibrium with the environment), and chemical nonflow exergy is the maximum work obtainable from a system as it is brought from the environmental state to the dead state (i.e., to complete equilibrium with the environment).

#### Exergy of Flows

#### Exergy of a Matter Flow

The exergy of a flowing stream of matter  $Ex_{flow}$  is the sum of nonflow exergy and the exergy associated with the flow work of the stream (with reference to  $P_0$ ), that is,

$$Ex_{flow} = Ex_{nonflow} + (P - P_0)V$$
(7)

or  $Ex_{flow}$  is expressed in terms of physical, chemical, kinetic, and potential components as

$$Ex_{flow} = Ex_{flow,ph} + Ex_0 + Ex_{kin} + Ex_{pot}$$
(8)

where

$$Ex_0 = \sum_i (\mu_{i0} - \mu_{i00}) N_i$$

$$Ex_{flow,ph} = (H - H_0) - T_0 (S - S_0)$$

#### Exergy of Thermal Energy

Consider a control mass, initially at the dead state, being heated or cooled at constant volume in an interaction with some other system. The heat transfer experienced by the control mass is Q. The flow of exergy associated with the heat transfer Q is denoted by  $Ex_Q$ , and is written as

$$\operatorname{Ex}_{Q} = \int_{i}^{t} (1 - T_{0}/T) \,\delta Q \tag{9}$$

where  $\delta Q$  is an incremental heat transfer, and the integral is from the initial state (*i*) to the final state (*f*). This 'thermal exergy' is the minimum work required by the combined system of the control mass and the environment in bringing the control mass to the final state from the dead state. If the temperature *T* of the control mass is constant, the thermal exergy transfer associated with a heat transfer is

$$Ex_{O} = (1 - T_{0}/T) Q = \tau Q$$
(10)

where  $\tau$  is called the 'exergetic temperature factor'.

#### Exergy of Work

The exergy associated with shaft work  $Ex_W$  is by definition  $W_x$ . The exergy transfer associated with work done by a system due to volume change is the net usable work due to the volume change, and is denoted by  $W_{\text{NET}}$ .

#### Exergy of Electricity

As for shaft work, the exergy associated with electricity is equal to the energy.

## Exergy Consumption (Destruction)

For a process occurring in a system, the difference between the total exergy flows into and out of the system, less the exergy accumulation in the system, is the exergy consumption (destruction)  $Ex_d$ , expressible as

$$Ex_d = T_0 S_{gen} \tag{11}$$

which points out that exergy destruction is proportional to entropy creation, and is known as the Gouy– Stodola relation.

#### **Reference Environment**

Exergy is evaluated with respect to a reference environment, so the intensive properties of the reference environment partly determine the exergy of a stream or system. The reference environment is in stable equilibrium, with all parts at rest relative to one another. No chemical reactions can occur between the environmental components. The reference environment acts as an infinite system, and is a sink and source for heat and materials. It experiences only internally reversible processes in which its intensive state remains unaltered (i.e., its temperature  $T_0$ , pressure  $P_0$ , and the chemical potentials  $\mu_{i00}$  for each of the *i* components present remain constant). The exergy of the reference environment is zero. The exergy of a stream or system is zero when it is in equilibrium with the reference environment.

#### **Energy and Exergy Efficiencies**

Efficiency has always been an important consideration in decision making regarding resource utilization and performance assessment of systems and applications. Efficiencies are often evaluated as ratios of energy quantities, and are often used to assess and compare various systems. Power plants, heaters, refrigerators, and thermal storages, for example, are often compared based on energy efficiencies or energybased measures of merit.

There are two key efficiencies as energy efficiency, based on energy analysis (under the first law of thermodynamics) and exergy efficiency (under the second law of thermodynamics). However, energy efficiencies are often misleading in that they do not always provide a measure of how nearly the performance of a system approaches ideality. Further, the thermodynamic losses which occur within a system (i.e., those factors which cause performance to deviate from ideality) often are not accurately identified and assessed with energy analysis. The results of energy analysis can indicate the main inefficiencies to be within the wrong sections of the system, and a state of technological efficiency different than actually exists. This requires a realistic efficiency to be defined. This can only be done by exergy efficiency since exergy analysis permits many of the shortcomings of energy analysis to be overcome. It, by stemming from the second law of thermodynamics, is useful in identifying the causes, locations and magnitudes of process inefficiencies. Efficiencies determined using ratios of exergy do provide a measure of an approach to an ideal. Exergy efficiencies are often more intuitively rational than energy efficiencies because efficiencies between 0% and 100% are always obtained. Measures which

can be greater than 100% when energy is considered, such as coefficient of performance, normally are between 0% and 100% when exergy is considered.

Energy  $(\eta_{en})$  and exergy  $(\eta_{ex})$  efficiencies are often written for steady-state processes occurring in systems as

$$\eta_{\rm en} = \frac{\text{Useful energy output}}{\text{Total energy input}}$$
(12)

$$\eta_{\rm ex} = \frac{\text{Useful exergy output}}{\text{Total exergy input}}$$
(13)

It is finally more important that exergy efficiencies often give more illuminating insights into process performance than energy efficiencies because (1) they weigh energy flows according to their exergy contents, and (2) they separate inefficiencies into those associated with effluent losses and those due to irreversibilities. In general, exergy efficiencies ultimately provide a true measure of potential for improvement.

### ILLUSTRATIVE EXAMPLE: ENERGY AND EXERGY ANALYSES OF INTEGRATED SYSTEMS

In this case study, energy and exergy analyses of two integrated systems are conducted to show how system performance differs from energy analysis to exergy analysis. First system studied is biomass gas cycle integrated with steam cycle and second system studied is geothermal flash system integrated with steam cycle. These two systems are studied to show the importance of renewable/alternative energy sources in future power generation and the importance of exergy analysis during designing stage of the system.

#### Systems Description Biomass Integrated with Steam Cycle

In this integrated system, biomass is used as a fuel in the combustion chamber. The integrated system studied is shown in Figure 2. Air is compressed from state 1 to state 2 using compressor whose power is supplied by the turbine. The compressed air at state 2 enters the combustion chamber where biomass is combusted with compressed air to leave at state 3 as exhaust gasses. Exhaust gasses enter the gas turbine where they expand to produce power. The exhaust gasses leaving gas turbine at state 4 are passed through the shell and tube heat exchanger where the release heat to the water entering the heat exchanger at state 7. After releasing heat in the heat exchanger, the exhaust gasses are released to the environment at state 5. The heat gained by the water entering the heat



**FIGURE 2** | Schematic of biomass cycle integrated with steam cycle.

exchanger leave at state 8 as a saturated vapor. The saturated vapor at state 8 enters the turbine where it expands to produce power. After expanding saturated mixture leaves at state 9 to enter the heat exchanger where it release heat to the air which is used as a cooling medium. After releasing heat condenser water enters the pump at state 6. The overall net power produced is used for later purposes.

# Geothermal Single Flash Integrated with Steam Cycle

In this system, geothermal water is used as an energy source. The integrated system studied is shown in Figure 3. The geothermal water at state 1 is expanded in the expansion valve to leave at state 2 as a saturated mixture. Saturated mixture at state 2 enters the separation chamber (flash chamber) where saturated vapor is extracted at state 3 and saturated water is extracted at state 6. Saturated vapor at state 3 enters the turbine where it expands to produce power and leaves at state 4. Stream at state 4 enters the condenser to release heat to air, which is used as cooling medium. After releasing heat, stream from state 4 leaves at state 5 where it combines with stream coming at state 7 to be reinjected to the geothermal well. Saturated water leaving at state 6 is passed through the shell and tube heat exchanger where it releases heat to the water coming at state 11. The heated water then enters the turbine at state 8, where it expands to produce power. After expanding, the vapor at state 9 enters the condenser to release heat to the air, which is used as cooling medium and leaves at state 10. Water leaving at state 10 enters the pump where its pressure is pumped to the pressure at state 11. The overall net power produced by the integrated system is used for later purposes.

## Analysis

## Biomass Integrated with Steam Cycle

First system studied consists of a gas cycle using biomass in a combustion chamber integrated with steam cycle for power production. The system studied is show in Figure 2. The topping cycle is a biomass gas-turbine cycle that has a pressure ratio of 8. Air enters the compressor (state 1) at 300 K and the turbine (state 3) at 1300 K. The isentropic efficiencies of the compressors, pumps and turbines are 85%. The bottoming cycle is a simple Rankine cycle operating between the pressure limits of 7 MPa and 5 kPa. The steam is heated to a temperature of 500°C



FIGURE 3 Schematic of geothermal cycle integrated with steam cycle.

in a heat exchanger by the heat supplied from the exhaust gases of a biomass combustion system. The exhaust gases leave the heat exchanger (state 5) at 450 K. The ambient temperature and pressure are assumed to be  $27^{\circ}$ C and 100 kPa, respectively. All the components are considered adiabatic. Biomass cycle is modeled on the basis of air-standard theorem. Air is treated as an ideal gas. It is assumed that 20% of the power is lost due to the parasitic losses.

#### *Energy and Exergy Analyses* The pressure at state 2 is calculated as

$$\Pr_2 = r_p \times \Pr_1 \tag{14}$$

where  $Pr_2$  is the reduced pressure at state 2,  $r_p$  is the pressure ratio, and  $Pr_1$  is the reduced pressure at state 1.

The pressure at state 4 is calculated as

$$\Pr_4 = \frac{1}{r_p} \times \Pr_3 \tag{15}$$

where  $Pr_4$  is the reduced pressure at state 4,  $r_p$  is the pressure ratio, and  $Pr_3$  is the reduced pressure at state 3.

The ideal specific work needed by compressor of biomass cycle to compress air from state 1 to state 2 is calculates as

$$w_{\rm comp,g,s} = h_{\rm s,2} - h_1$$
 (16)

where  $w_{\text{comp,g,s}}$  is the ideal specific work needed by the compressor of the biomass cycle,  $h_{s,2}$  is the ideal specific enthalpy at state 2 and  $h_1$  is the specific enthalpy at state 1.

The ideal specific power produced by turbine of the biomass cycle is defined as

$$w_{\text{turb},g,s} = h_3 - h_{s,4}$$
 (17)

where  $w_{\text{turb,g,s}}$  is the ideal specific work produced by the turbine of the biomass cycle,  $h_3$  is the specific enthalpy at state 3, and  $h_{s,4}$  is the ideal specific enthalpy at state 4.

The ideal specific heat input to the biomass cycle is found using

$$q_{\rm in,g,s} = h_3 - h_{\rm s,2} \tag{18}$$

where  $q_{in,g,s}$  is the ideal specific heat input to the combustion chamber of the biomass cycle,  $h_3$  is the specific enthalpy at state 3 and  $h_{s,2}$  is the ideal specific enthalpy at state 2.

The ideal parasitic loss is calculated as

$$w_{\text{parasitic,g,s}} = 0.2 \times (w_{\text{turb,g,s}} - w_{\text{comp,g,s}})$$
(19)

where  $w_{\text{parasitic,g,s}}$  is the ideal specific parasitic loss of the biomass cycle.

The ideal specific net power produced by the biomass cycle is defined as

$$w_{\text{net,g,s}} = w_{\text{turb,g,s}} - w_{\text{comp,g,s}} - w_{\text{parasitic,g,s}}$$
(20)

where  $w_{\text{net,g,s}}$  is the ideal net specific power produced by the biomass cycle.

The specific exergy at state 1 is found using

$$ex_1 = (h_1 - h_0) - T_0 \times (s_1 - s_0)$$
(21)

where  $ex_1$  represent specific exergy at state 1,  $h_0$  represents specific enthalpy at ambient state,  $s_1$  is specific entropy at state 1, and  $s_0$  is specific entropy at ambient state.

The same formulation is used to caculate specific exergy at each state. The ideal specific thermal exergy of the biomass cycle is found using

$$\operatorname{ex}_{\operatorname{th},\mathrm{g},\mathrm{s}} = \left[1 - \frac{T_0}{T_{\operatorname{avg},\mathrm{g},\mathrm{s}}}\right] \times q_{\operatorname{in},\mathrm{g},\mathrm{s}}$$
(22)

where  $T_{\text{avg,g,s}} = \frac{(T_3 + T_{s,2})}{2}$  and  $\text{ex}_{\text{th,g,s}}$  is the ideal specific thermal exergy of the biomass cycle, and  $T_{\text{avg,g,s}}$  is the ideal average temperature.

The ideal energy and exergy efficiencies of the biomass cycle are calculated using

$$\eta_{\rm en,g,s} = \frac{w_{\rm net,g,s}}{q_{\rm in,g,s}} \tag{23}$$

$$\eta_{\rm ex,g,s} = \frac{w_{\rm net,g,s}}{e x_{\rm th,g,s}}$$
(24)

The actual specific work needed by compressor of biomass cycle to compress air from state 1 to state 2 is calculates as

$$w_{\rm comp,g,a} = \frac{h_{\rm s,2} - h_1}{\eta} \tag{25}$$

where  $w_{\text{comp,g,a}}$  is the actual specific work needed by the compressor of the biomass cycle,  $h_{s,2}$  is the ideal specific enthalpy at state 2,  $h_1$  is the specific enthalpy at state 1, and  $\eta$  is the isentropic efficiency which is 85%.

The actual specific power produced by turbine of the biomass cycle is defined as

$$w_{\text{turb},g,a} = \eta \times (h_3 - h_{s,4}) \tag{26}$$

where  $w_{\text{turb,g,a}}$  is the actual specific work produced by the turbine of the biomass cycle,  $h_3$  is the specific enthalpy at state 3,  $h_{\text{s},4}$  is the ideal specific enthalpy at state 4, and  $\eta$  is the isentropic efficiency which is 85%.

The actual specific heat input to the biomass cycle is found using

$$q_{\rm in,g,a} = h_3 - h_2$$
 (27)

where  $q_{\text{in},\text{g},\text{a}}$  is the actual specific heat input to the combustion chamber of the biomass cycle,  $h_3$  is the specific enthalpy at state 3, and  $h_2$  is the actual specific enthalpy at state 2.

The actual parasitic loss is calculated as

$$w_{\text{parasitic,g,a}} = 0.2 \times (w_{\text{turb,g,a}} - w_{\text{comp,g,a}})$$
 (28)

where  $w_{\text{parasitic,g,a}}$  is the actual specific parasitic loss of the biomass cycle.

The actual specific net power produced by the biomass cycle is defined as

$$w_{\text{net,g,a}} = w_{\text{turb,g,a}} - w_{\text{comp,g,a}} - w_{\text{parasitic,g,a}}$$
 (29)

where  $w_{\text{net,g,a}}$  is the actual net specific power produced by the biomass cycle.

The actual specific thermal exergy of the biomass cycle is found using

$$\operatorname{ex}_{\operatorname{th},\mathrm{g},\mathrm{a}} = \left[1 - \frac{T_0}{T_{\operatorname{avg},\mathrm{g},\mathrm{a}}}\right] \times q_{\operatorname{in},\mathrm{g},\mathrm{a}} \tag{30}$$

where  $T_{\text{avg,g,a}} = \frac{(T_3 + T_{a,2})}{2}$  and  $\text{ex}_{\text{th,g,a}}$  is the actual specific thermal exergy of the biomass cycle, and  $T_{\text{avg,g,a}}$  is the actual average temperature.

The actual energy and exergy efficiencies of the biomass cycle are calculated using

$$\eta_{\rm en,g,a} = \frac{w_{\rm net,g,a}}{q_{\rm in,g,a}} \tag{31}$$

$$\eta_{\rm ex,g,s} = \frac{w_{\rm net,g,a}}{e x_{\rm th,g,a}} \tag{32}$$

The specific power consumed by pump of the steam cycle is found using

$$w_{\rm p} = v_6 \left[ \frac{P_7 - P_6}{\eta} \right] \tag{33}$$

where  $w_p$  is specific power consumed by the pump,  $v_6$  is specific volume at state 6, and  $\eta$  is the isentropic efficiency of the pump.

The mass ratio 'y' is defined as mass flow rate of the steam divided by mass flow rate of the biomass gas and is calculated using

$$y = \frac{h_4 - h_5}{h_8 - h_7} \tag{34}$$

The specific power produced by turbine of the steam cycle is defined as

$$w_{\text{turb,st}} = (h_8 - h_9)$$
 (35)

where  $w_{turb,st}$  is the specific work produced by the turbine of the steam cycle,  $h_8$  is the specific enthalpy at state 8, and  $h_9$  is the specific enthalpy at state 9.

The specific heat input to the steam cycle is found using

$$q_{\rm in,st} = h_8 - h_7$$
 (36)

where  $q_{\text{in,st}}$  is the specific heat input to the steam cycle,  $h_8$  is the specific enthalpy at state 8, and  $h_7$  is the specific enthalpy at state 7.

The specific heat ouput from the condenser of the steam cycle is found using

$$q_{\rm con,st} = h_9 - h_6 \tag{37}$$

where  $q_{\text{con,st}}$  represents specific heat output from the condenser of the steam cycle.

The parasitic loss is calculated as

$$w_{\text{parasitic,st}} = 0.2 \times (w_{\text{turb,st}} - w_{\text{p}})$$
 (38)

where  $w_{\text{parasitic,st}}$  is the specific parasitic loss of the steam cycle.

The specific net power produced by the steam cycle is defined as

$$w_{\rm net,st} = w_{\rm turb,st} - w_{\rm p} - w_{\rm parasitic,st}$$
(39)

where  $w_{\text{net,g,a}}$  is the actual net specific power produced by the biomass cycle.

The specific input thermal exergy of the steam cycle is found using

$$ex_{th,in,st} = \left[1 - \frac{T_o}{T_{avg,st}}\right] \times q_{in,st}$$
(40)

where  $T_{\text{avg,st}} = \frac{(T_7 + T_8)}{2}$  and  $\text{ex}_{\text{th,in,st}}$  is the specific input thermal exergy of the steam cycle, and  $T_{\text{avg,st}}$  is the average temperature.

The specific condenser thermal exergy of the steam cycle is found using

$$ex_{th,con,st} = \left[1 - \frac{T_o}{T_{avg,con}}\right] \times q_{con,st}$$
(41)

where  $T_{\text{avg,con}} = \frac{(T_9+T_6)}{2}$  and  $\text{ex}_{\text{th,con,st}}$  is the specific condenser thermal exergy of the steam cycle, and  $T_{\text{avg,con}}$  is the average temperature.

The energy and exergy efficiencies of the steam cycle are calculated using

$$\eta_{\rm en,st} = \frac{w_{\rm net,st}}{q_{\rm in,st}} \tag{42}$$

$$\eta_{\rm ex,g,s} = \frac{w_{\rm net,st}}{ex_{\rm th,st}} \tag{43}$$

The net overall power output of the integrated system is found using

$$w_{\rm net,ov} = y \times w_{\rm net,st} - w_{\rm net,g,a}$$

The ideal specific exergy destruction in the compressor is found using

$$ex_{dest,comp,g,s} = ex_1 - ex_{s,2} + w_{comp,g,s}$$
(44)

where  $ex_{dest,comp,g,s}$  represents ideal specific exergy destruction in the compressor.

The actual specific exergy destruction in the compressor is found using

$$ex_{dest,comp,g,a} = ex_1 - ex_2 + w_{comp,g,a}$$
(45)

where  $ex_{dest,comp,g,a}$  represents actual specific exergy destruction in the compressor.

The ideal specific exergy destruction in the combustion chamber is found using

$$ex_{dest,cc,g,s} = ex_{s,2} - ex_3 + ex_{th,g,s}$$
(46)

where  $ex_{dest,cc,g,s}$  represents ideal specific exergy destruction in the combustion chamber.

The actual specific exergy destruction in the combustion chamber is found using

$$ex_{dest,cc,g,a} = ex_2 - ex_3 + ex_{th,g,a}$$
(47)

where  $ex_{dest,cc,g,a}$  represents actual specific exergy destruction in the combustion chamber.

The ideal specific exergy destruction in the turbine of the biomass cycle is found using

$$ex_{dest,turb,g,s} = ex_3 - ex_{s,4} - w_{turb,g,s}$$
(48)

where  $ex_{dest,turb,g,s}$  represents ideal specific exergy destruction in the turbine of the biomass cycle.

The actual specific exergy destruction in the turbine of the biomass cycle is found using

$$ex_{dest,turb,g,a} = ex_3 - ex_4 - w_{turb,g,a}$$
(49)

where  $ex_{dest,turb,g,a}$  represents actual specific exergy destruction in the turbine of the biomass cycle.

The specific exergy destruction in the turbine of the steam cycle is found using

$$\kappa_{\text{dest,turb,st}} = ex_8 - ex_9 - w_{\text{turb,st}}$$
(50)

where  $ex_{dest,turb,st}$  represents specific exergy destruction in the turbine of the steam cycle.

The specific exergy destruction in the pump of the steam cycle is found using

$$ex_{dest,p,st} = ex_6 - ex_7 + w_p \tag{51}$$

where  $ex_{dest,p,st}$  represents specific exergy destruction in the pump of the steam cycle.

The specific exergy destruction in the heat exchange is found using

$$ex_{dest,he} = ex_4 + ex_7 - ex_5 - ex_8$$
 (52)

where  $ex_{dest,he}$  represents specific exergy destruction in the heat exchanger.

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The specific exergy destruction in the condenser of the steam cycle is found using

$$ex_{dest,con,st} = ex_9 - ex_6 - ex_{th,con,st}$$
(53)

where  $ex_{dest,con,st}$  represents specific exergy destruction in the condenser of the steam cycle.

The overall specific exergy destruction is found using

$$ex_{dest,ov} = ex_1 + ex_{th,g,a} + w_p + w_{comp,g,a} - ex_5$$
$$-w_{turb,g,a} - w_{turb,st} - ex_{th,in,st}$$
(54)

where  $ex_{dest,ov}$  represents overall specific exergy destruction.

The overall energy and exergy efficiencies are then defined as

$$\eta_{\rm en,ov} = \frac{w_{\rm net,ov}}{q_{\rm in,g,a}} \tag{55}$$

$$\eta_{\rm ex,ov} = \frac{w_{\rm net,ov}}{e x_{\rm th,g,a}}$$
(56)

# Geothermal Single Flash Integrated with Steam Cycle

The second system studied here consists of a geothermal single flash integrated with steam cycle for power production. The system studied is show in Figure 3. The enthalpy of water as it enters to the system (state 1) is 2255 kJ/kg. The associated mass flow rate is determined to be 100 kg/second at state 1. The fluid at state 8 is saturated water vapor. The ambient temperature and pressure are at 25°C and 100 kPa. The turbine and pump efficiencies are 0.85. The mass flow rate of steam cycle is taken to be 6 kg/second. All the components are considered adiabatic. It is assumed that 20% of the power is lost due to the parasitic losses.

#### Energy and Exergy Analyses

The specific power consumed by pump of the steam cycle is found using

$$w_{\rm p} = v_{10} \left[ \frac{P_{11} - P_{10}}{\eta} \right] \tag{57}$$

where  $w_p$  is specific power consumed by the pump,  $v_{10}$  is specific volume at state 6, and  $\eta$  is the isentropic efficiency of the pump.

The mass flow rates at state 3 and state 6 are found using

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_6$$
 (58)

$$\dot{m}_2 x_2 = \dot{m}_3 x_3 + \dot{m}_6 x_6 \tag{59}$$

where  $\dot{m}_2$  is the mass flow rate at state 2,  $\dot{m}_3$  is mass flow rate at state 3,  $\dot{m}_6$  is mass flow rate at state 6,  $x_2$ is quality at state 2,  $x_3$  is quality at state 3, and  $x_6$  is quality at state 6.

The condenser load of flash power plant is found using

$$\dot{Q}_{\rm con,f} = \dot{m}_4 h_4 - \dot{m}_5 h_5 \tag{60}$$

where  $\dot{Q}_{con,f}$  represents condenser load of the flash power plant.

The condenser load of steam power plant is found using

$$\dot{Q}_{\rm con,st} = \dot{m}_9 h_9 - \dot{m}_{10} h_{10} \tag{61}$$

where  $\dot{Q}_{con,st}$  represents condenser load of the steam power plant.

The power produced by the turbine of flash power plant is calculated using

$$\dot{W}_{t,f} = \dot{m}_3 h_3 - \dot{m}_4 h_4$$
 (62)

where  $\dot{W}_{t,f}$  represents power produced by the turbine of the flash power plant.

The power produced by the turbine of steam power plant is calculated using

$$\dot{W}_{t,st} = \dot{m}_8 h_8 - \dot{m}_9 h_9$$
 (63)

where  $\dot{W}_{t,st}$  represents power produced by the turbine of the steam power plant.

The consumed by the pump is found using

$$\dot{W}_{\rm p} = \dot{m}_{11} h_{11} - \dot{m}_{10} h_{10} \tag{64}$$

where  $\dot{W}_p$  represents power consumed by the pump. The overall parasitic loss is calculated as

$$\dot{W}_{\text{parasitic,ov}} = 0.2 \times (\dot{W}_{\text{t,st}} + \dot{W}_{\text{t,f}} - \dot{W}_{\text{p}}) \tag{65}$$

where  $\dot{W}_{parasitic,ov}$  is the parasitic loss of the steam cycle.

The net power produced by the overall system is found using

$$\dot{W}_{\text{net,ov}} = \dot{W}_{\text{t,st}} + \dot{W}_{\text{t,f}} - \dot{W}_{\text{p}} - \dot{W}_{\text{parasitic,ov}}$$
(66)

where  $\dot{W}_{net,ov}$  is the net power produced by the overall system.

The thermal exergy rate of the condenser of the flash power plant is found using

$$\dot{\text{ex}}_{\text{con,th,f}} = \left[1 - \frac{T_0}{T_{\text{avg,con,f}}}\right] \times \dot{Q}_{\text{con,f}}$$
(67)

where  $T_{\text{avg,con,f}} = \frac{(T_4+T_5)}{2}$  and  $ex_{\text{con,th,f}}$  is the thermal exergy rate of the condenser of the flash power plant, and  $T_{\text{avg,con,f}}$  is the average temperature of the condenser of the flash power plant.

The thermal exergy rate of the condenser of the steam power plant is found using

$$\dot{\text{ex}}_{\text{con,th,st}} = \left[1 - \frac{T_0}{T_{\text{avg,con,st}}}\right] \times \dot{Q}_{\text{con,st}}$$
 (68)

where  $T_{\text{avg,con,st}} = \frac{(T_9+T_{10})}{2}$  and  $ex_{\text{con,th,st}}$  is the thermal exergy rate of the condenser of the steam power plant, and  $T_{\text{avg,con,st}}$  is the average temperature of the condenser of the steam power plant.

The exergy destruction rate in the flash chamber is found using

$$\dot{ex}_{dest,flashchamber} = \dot{ex}_2 - \dot{ex}_3 - \dot{ex}_6 \tag{69}$$

where  $\dot{ex}_2 = \dot{m}_2 [(h_2 - h_0) - T_0 (s_2 - s_0)]$  and  $\dot{ex}_{dest,flashchamber}$  represents exergy destruction rate in the flash chamber,  $\dot{ex}_2$  represents exergy rate at state 2,  $\dot{ex}_3$  represents exergy rate at state 3, and  $\dot{ex}_6$  represents exergy rate at state 6. The rate of exergy at each state is found using same formula as mentioned above.

The exergy destruction rate in the turbine of the flash power plant is found using

$$\dot{ex}_{dest,t,f} = \dot{ex}_3 - \dot{ex}_4 - \dot{W}_{t,f}$$
(70)

where  $\dot{ex}_{dest,t,f}$  represents exergy destruction rate in the turbine of the flash power plant.

The exergy destruction rate in the condenser of the flash power plant is found using

$$\dot{ex}_{dest,con,f} = \dot{ex}_4 - \dot{ex}_5 - \dot{ex}_{con,th,f}$$
(71)

where  $\dot{ex}_{dest, con, f}$  represents exergy destruction rate in the condenser of the flash power plant.

The exergy destruction rate in the turbine of the steam power plant is found using

$$\dot{ex}_{dest,t,st} = \dot{ex}_8 - \dot{ex}_9 - \dot{W}_{t,st}$$
(72)

where  $\dot{ex}_{dest,t,st}$  represents exergy destruction rate in the turbine of the steam power plant.

The exergy destruction rate in the condenser of the steam power plant is found using

$$\dot{ex}_{dest,con,st} = \dot{ex}_9 - \dot{ex}_{10} - \dot{ex}_{con,th,st}$$
(73)

where  $e\dot{x}_{dest, con, st}$  represents exergy destruction rate in the condenser of the steam power plant.

The exergy destruction rate in the pump is calculated using

$$\dot{ex}_{dest,p} = \dot{ex}_{10} - \dot{ex}_{11} + \dot{W}_{p}$$
 (74)

where  $e\dot{x}_{dest,p}$  represents exergy destruction rate in the pump.

The overall exergy destruction rate in the integrated system is defined as

$$\begin{aligned} s_{t,ov} &= e\dot{\mathbf{x}}_1 + \dot{W}_p - e\dot{\mathbf{x}}_{12} - \dot{W}_{t,f} \\ &- \dot{W}_{t,st} - e\dot{\mathbf{x}}_{con,th,f} - e\dot{\mathbf{x}}_{con,th,st} \end{aligned} \tag{75}$$

The overall energy and exergy efficiencies are the defined as

$$\eta_{\rm en,ov} = \frac{\dot{W}_{\rm net,ov}}{(\dot{m}_1 h_1 - \dot{m}_{12} h_{12})}$$
(76)

$$\eta_{\rm ex,ov} = \frac{\dot{W}_{\rm net,ov}}{(\dot{ex}_1 - \dot{ex}_{12})}$$
(77)

#### Results and Discussion Biomass Integrated with Steam Cycle

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Energy and exergy analyses are carried out for each component of the integrated system. Conducting exergy analysis is very important to see how system will perform in real life. The results obtained are shown in the Tables 1 and 2.

The results shown in Table 2 indicated that system performance vary from energy analysis to exergy analysis. The results show that for the given ambient condition the exergetic efficiencies of the biomass cycle, steam cycle, and overall system are higher than energetic efficiencies of the respective systems. It is observed that for the ideal case of biomass cycle when isentropic efficiencies of the compressor and turbine are taken to be 100%, energetic and exergetic efficiencies are 34.06, and 50.38%, respectively. However, for the actual case of biomass cycle when isentropic efficiencies of the compressor and turbine are taken to be 85%, the energetic and exergetic efficiencies are found to be 22.6 and 33.08%, respectively. However, for the steam cycle energetic and exergetic efficiencies are found to be 27.67 and 61.77%, respectively. Moreover, for the overall system energetic and exergetic efficiencies turn out to be 37.26 and 54.53%, respectively. These results show that efficiencies vary from energy analysis to exergy analysis. In this case, exergy efficiencies are obtained to be higher than energy efficiencies because of the ambient conditions chosen. It is not true to generalize that exergy efficiencies are always lower than energy efficiencies. As it can be seen in this case, exergy efficiencies can be higher than energy efficiencies. Figure 4 displays specific exergy destruction in each component of the system studied and in the overall system. It can be seen that the maximum specific exergy is destructed in heat exchanger. This shows that heat exchanger

State Points	Specific Exergy (kJ/kg)	Specific Enthalpy (kJ/kg)	Reduced Pressure (kPa)	Pressure (kPa)	Specific Entropy (kJ/kg K)	Temperature (K)	Specific Volume (m³/kg)
0	N/A	N/A	N/A	100	N/A	298	N/A
1	12.42	300.4	1.386	N/A	1.702	300	N/A
2s	53.9	544.7	11.09	N/A	2.299	540	N/A
2a	74.67	587.8	11.09	N/A	2.374	582	N/A
3	615	1396	330.9	N/A	3.273	1300	N/A
4s	186.3	789.4	41.36	N/A	2.676	770	N/A
4a	244.8	880.4	41.36	N/A	2.785	853	N/A
5	15.86	452.1	5.775	N/A	2.116	450	N/A
6	0.3484	137.8	N/A	5	0.4762	306	0.001005
7	7.404	146	N/A	7000	0.4803	306.5	N/A
8	1389	3411	N/A	7000	6.8	773	N/A
9	56.35	2274	N/A	5	7.455	306	N/A

TABLE 1	Thermodynamic Propertie	s at Each State Point for Integrated	Biomass and Steam Cycle System
		5	, , ,

 TABLE 2 | Results of the Integrated Biomass and Steam Cycle System

Components	Specific Energy (kJ/kg)	Specific Thermal Exergy (kJ/kg)	Specific Exergy Destruction (kJ/kg)	Energy Efficiency (%)	Exergy Efficiency (%)
Biomass cycle compressor (ideal)	244.2	N/A	202.8	N/A	N/A
Biomass cycle compressor (actual)	287.9	N/A	225.1	N/A	N/A
Biomass cycle combustion chamber (ideal)	851.4	575.6	14.58	N/A	N/A
Biomass cycle combustion chamber (actual)	808.3	552.4	12.05	N/A	N/A
Biomass cycle turbine (ideal)	606.7	N/A	178.1	N/A	N/A
Biomass cycle turbine (actual)	515.7	N/A	145.6	N/A	N/A
Steam cycle turbine	1137	N/A	195.5	N/A	N/A
Steam cycle pump	8.273	N/A	1.218	N/A	N/A
Heat exchanger	3265	1462	1153	N/A	N/A
Steam cycle condenser	2136	56	0	N/A	N/A
Steal cycle net	903.4	N/A	N/A	N/A	N/A
Biomass cycle net (ideal)	290	N/A	N/A	N/A	N/A
Biomass cycle net (actual)	182.7	N/A	N/A	N/A	N/A
Biomass cycle (ideal)	N/A	N/A	N/A	34.06	50.38
Biomass cycle (actual)	N/A	N/A	N/A	22.6	33.08
Steam cycle	N/A	N/A	N/A	27.67	61.77
Overall system	301.2	N/A	2271	37.26	54.53

can be modified to reduce the exergy destruction in it so that more power can be generated. Figure 5 helps visualizing how overall energy and exergy efficiencies vary with variation in ambient temperature. It is observed that the overall energy efficiency does not vary with rise in ambient temperature but the overall exergy efficiency increases from 53.9 to 55.6% with rise in ambient temperature.

# Geothermal Single Flash Integrated with Steam Cycle

Energy and exergy analyses are carried out for each component of the integrated system. Conducting ex-

ergy analysis is very important to study how system will perform in real life. The results obtained from this study are tabulated in the Tables 3 and 4.

The results obtained show case that system performances differ a lot from energy analysis to exergy analysis. It is also important to conduct exergy analysis as it shows where most of exergy is being destructed. It can be seen in Table 4 that most amount of exergy is being destructed in the turbine of the geothermal cycle followed by condenser of the geothermal cycle. Conducting exergy analysis shows that more can be produced from the same geothermal turbine if the exergy destruction is minimized. Finding where most of energy is lost is not possible



**FIGURE** 4 | Specific exergy destruction in each component and overall system.



FIGURE 5 | Variation in overall energy and exergy efficiencies with rise in ambient temperature.

by conducting only energy analysis, it is exergy analysis which pin points where maximum amount of exergy is being lost. Also, the overall energy and exergy efficiencies obtained are 16.67 and 44.3%, respectively. This exergy analysis is conducted for a specific ambient condition and it is important to mention that exergetic parameters vary with variation in ambient conditions. Figure 6 shows how much exergy destruction rate takes place in each component of the system and in overall system. It can be seen that

State Points	Exergy Rate (kW)	Specific Enthalpy (kJ/kg)	Mass Flow Rate (kg/second)	Pressure (kPa)	Specific Entropy (kJ/kg K)	Temperature (K)	Quality	Specific Volume (m³/kg)
0	N/A	104.8	N/A	100	0.3669	298.2	N/A	N/A
1	81305	2255	100	8243	4.852	570.3	0.65	N/A
2	61749	2255	100	791.5	5.508	443.2	0.7494	N/A
3	58862	2768	74.94	791.5	6.666	443.1	1	N/A
4	10068	2210	74.94	10	6.976	318.9	0.8436	N/A
5	6.438	108.1	74.94	10	0.378	298.9	N/A	N/A
6	2881	719.2	25.06	791.5	2.042	443.1	0	N/A
7	17.49	108.9	25.06	791.5	0.3783	299	N/A	N/A
8	4164	2741	6	420	6.88	418.5	1	N/A
9	827.1	2263	6	10	7.144	318.9	0.8661	N/A
10	16.84	191.7	6	10	0.6489	318.9	0	0.00101
11	19.36	192.2	6	420	0.6491	319	N/A	N/A
12	8.53	108.3	100	10	0.3787	299	N/A	N/A

TABLE 3	Thermodynamic Properties	at Each State Point for Inte	egrated Geothermal	Single Flash and Stean	n Cvcle Svstem
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 TABLE 4 | Results of the Integrated Geothermal Single Flash and Steam Cycle System

Components	Energy Rate (kW)	Thermal Exergy Rate (kW)	Exergy Destruction Rate (kW)	Energy Efficiency (%)	Exergy Efficiency (%)
Geothermal cycle condenser	157498	5501	4561	N/A	N/A
Geothermal cycle turbine	41879	N/A	6915	N/A	N/A
Steam cycle condenser	12430	810.2	0	N/A	N/A
Steam cycle turbine	2864	N/A	473	N/A	N/A
Steam cycle pump	2.924	N/A	0.41	N/A	N/A
Heat exchanger	N/A	N/A	1281	N/A	N/A
Flash chamber	N/A	N/A	5.763	N/A	N/A
Overall system	35792	N/A	30246	16.67	44.3

maximum exergy destruction rate takes place in the turbine of the geothermal cycle. This is a sign that if one needs to improve the performance of the overall system, the performance of the geothermal cycle turbine should get first preference in component improvement strategy. Variation in the overall energetic and exergetic efficiencies with rise in ambient temperature is plotted in Figure 7. It can be seen that overall energetic efficiency remains constant and is not affected by rise in ambient temperature. However, overall exergetic efficiency increases from 42.1 to 47.1% with rise in ambient temperature.

## CONCLUSIONS

In this article, we discuss the role of exergy and its use for a new concept of AIDA. A general exergy analysis methodology is presented along with both energy and exergy efficiencies. Two illustrative examples, studying two integrated systems, are presented,

which involve energy and exergy analyses, and energy and exergy efficiency assessments of two environmentally benign integrated power generating sources. Exergy destruction rates for the components of the system considered are investigated for possible improvement, and some parametric studies are also performed to rank these components accordingly. Furthermore, it is obtained that when ambient temperature is varied the energy efficiency of the overall system in both the case studies remains constant whereas exergy efficiency of the overall system in both the case studies varies vastly. The present study show that energy analysis itself is not sufficient enough to determine performance of any system and exergy analysis should always be considered to project more appropriate performance of a system. It is thus concluded that in first case study, heat exchanger destructs the highest amount of exergy (1153 kJ/kg) and in second case study, geothermal cycle turbine destructs a maximum amount of exergy (6915 kW).



**FIGURE 6** | Exergy destruction in each component and overall system.



FIGURE 7 | Variation in overall energy and exergy efficiencies with rise in ambient temperature.

## NOMENCLATURE

h = specific enthalpy (kJ/kg)  $A = area (m^2)$ H = enthalpy (kJ)e = specific energy (kJ/kg)ke = specific kinetic energy (kJ/kg) E = energy; total solar energy reaching solar KE = kinetic energy (kJ)pond; wind energy (kW) ex = specific exergy (kJ/kg)m = mass (kg)  $\dot{m} = \text{mass flow rate (kg/second)}$  $\dot{Ex} = exergy rate (kW)$ P = pressure (kPa)Ex = exergy (kJ)Pr = reduced pressure (kPa) $Ex_Q$  = exergy associated with heat Q (kJ)

pe = specific potential energy, kJ/kg  $\tau =$  exergetic temperature factor PE = potential energy (kI) $\psi = \text{exergy efficiency}$ q = specific heat transfer (kJ/kg) O = heat, kISubscripts  $\dot{Q}$  = heat transfer rate (kW) a = actual $Q_r$  = heat transfer into system avg = averageacross region r on system cc = combustion chamberboundary (kJ) comp = compressor $r_{\rm p} = {\rm pressure \ ratio}$ con = condensers =specific entropy (kJ/kg K) d = destructionS = entropy (kJ)dest = destruction $S_{\text{gen}} = \text{entropy generation } (kJ)$ en = energy $\Delta S = \text{entropy change (kJ)}$ ex = exergyT =temperature (K) f = flashu = specific internal energy (kJ/kg) g = biomass gasU = internal energy; heat loss from pond surface gen = generationto air (kI) he = heat exchanger $v = \text{specific volume } (\text{m}^3/\text{kg})$ kin = kinetic $\mathbf{V} = \text{volume} (\text{m}^3)$ ov = overallV = velocity (m/second) p = pumpw = specific work (kW/kg)ph = physical $\dot{W} =$ work rate (kW) pot = potential x = qualityQ = heaty = mass ratios = idealst = steam**Greek Letters** t = turbine $\eta = efficiency$ th = thermal $\mu$  = chemical potential turb = turbine  $\Pi = entropy creation$ 1-11 = state numbers $\rho = \text{density}$ 0 = reference-environment state

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